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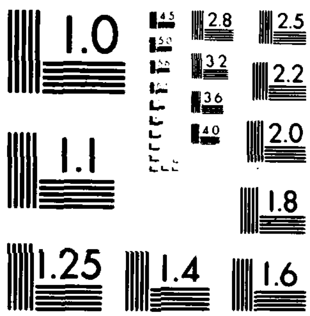

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Boiling Heat Transfer in Confined Space

Annual Technical Report
September 1980

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Associate Professor
Department of Mechanical Engineering
Carnegie-Mellon University
Pittsburgh, PA 15213

Prepared for
M.K. Ellingsworth, Program Monitor
The Office of Naval Research
Arlington, VA 22217

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In many equipments, boiling occurs in confined space such as the clearence between the tube and the support plate of steam generators. Corrosive concentration builds up at the boundary of dryout zone and induces severe damage. The knowledge on this kind of boiling phenomena is very limited. It is the objective of this research to understand this fundamental heat transfer of this problem through systematical analysis and experimental studies.
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This report describes mainly the analysis performed in the past year at Carnegie-Mellon University. The forced convective two phase flow and boiling heat transfer in confined space is studied using the subchannel analysis which is an approximated method but effective in two phase calculations. Reasonable prediction of the dryout phenomena has been achieved. A separate analysis is performed for single phase flow heat transfer. This differential analysis is formulated from the lubrication theory and calculations are performed for both the line-contact or two-point contact configurations. Finally, the progress in experimental research is also reported here.

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September 1980

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TABLE OF CONTENTS

SUMMARY

In many equipments, boiling occurs in confined space such as the clearance between the tube and the support plate of steam generators. Corrosive concentration builds up at the boundary of dryout zone and induces severe damage. The knowledge on this kind of boiling phenomena is very limited. It is the objective of this research to understand this fundamental heat transfer of this problem through systematical analysis and experimental studies.

This report describes mainly the analysis performed in the past year at Carnegie-Mellon University. The forced convective two phase flow and boiling heat transfer in confined space is studied using the subchannel analysis which is an approximated method but effective in two phase calculations. Reasonable prediction of the dryout phenomena has been achieved. A separate analysis is performed for single phase flow heat transfer. This differential analysis is formulated from the lubrication theory and calculations are performed for both the line-contact and two-point-contact configurations. Finally, the progress in experimental research is also reported here.

CONCLUSION

Both the subchannel analysis and the differential analysis calculate the flow field in confined space satisfactorily. At single phase flow the differential analysis is recommended. At two phase flow the subchannel analysis is very effective.

The dry-out pattern and the temperature field in the confined space have been predicted by the subchannel analysis for the condition of line-contact. The results are reasonable and encouraging.

The fluid flow and heat transfer at single phase laminar flow have been analyzed by the differential analysis. The formulation and the results are presented in non-dimensional form for general applications.

In the coming year, the experimental data will be obtained to validate the analysis.

I. INTRODUCTION

Boiling at conventional heat transfer surface has been studied extensively in the past thirty years [1]. However, at the same time, the boiling in confined space has been almost completely neglected. The existing information [2] [3], is limited and incomplete.

In many equipments, boiling occurs in confined space. For example, in the steam generators, clearance is allowed between the heated tube and its support-plate where the tube runs through. In the clearance, flow is reduced and boiling occurs. Another example occurs in nuclear reactor core where the structures are heated by nuclear radiation. Boiling occurs at the narrow space among the structures.

The boiling of liquid in confined space leads to permanent dryout. Corrosive concentration may build up at the boundary of dryout zone where boiling occurs. Solid deposits may also occur at the dryout boundary.

The understanding of the thermal-hydraulic related corrosion in confined space is lacking. This is primary because the boiling heat transfer in confined space is not really known. The fundamental understanding of the boiling heat transfer in confined space is also important to the advances in power engineering and lubrication engineering.

In view of this need, a systematic study of boiling heat transfer in confined space is performed at the Dept. of Mechanical Engineering of Carnegie-Mellon University under the support of the Office of Naval Research. Both experimental and analytical research are performed. The experimental result of concentric annulus provides the fundamental information on boiling heat transfer in confined space. The dryout pattern in eccentric annulus will be used to validate the prediction of the analysis.

In this first annual report, the theoretical analysis and its preliminary results will be presented in detail. The current status of the test loop and test section will also be reported.

II. ANALYSIS

Fluid flow and heat transfer are analyzed for the narrow confined space between a tube and the tube support plate. The tube can be either inclined or parallel to the hole of the support plate. When they are parallel, the annulus may be concentric or eccentric. Two practically important extreme conditions of line-contact and two-point-contact are considered as shown in Figure 1. Due to the nature of symmetry, only 180^{0} of the total confined space need be studied. Additionally, the curvature of the channel can be neglected in analysis, such that, the annulus can be analyzed as a flat channel with varying channel thickness. The typical case of two-point-contact is shown in Figure 2(a) and the case of line-contact in Figure 2(b).

Generally, the gap thickness h for an eccentric annulus, see Figure 1(b), can be expressed acurately, but not exactly, by

$$h = c\,(1 - \varepsilon \cos \theta) \tag{1}$$

where c is the average thickness. ε is e/c where the e is defined in Figure 1(b).

Therefore, the gap thickness h is

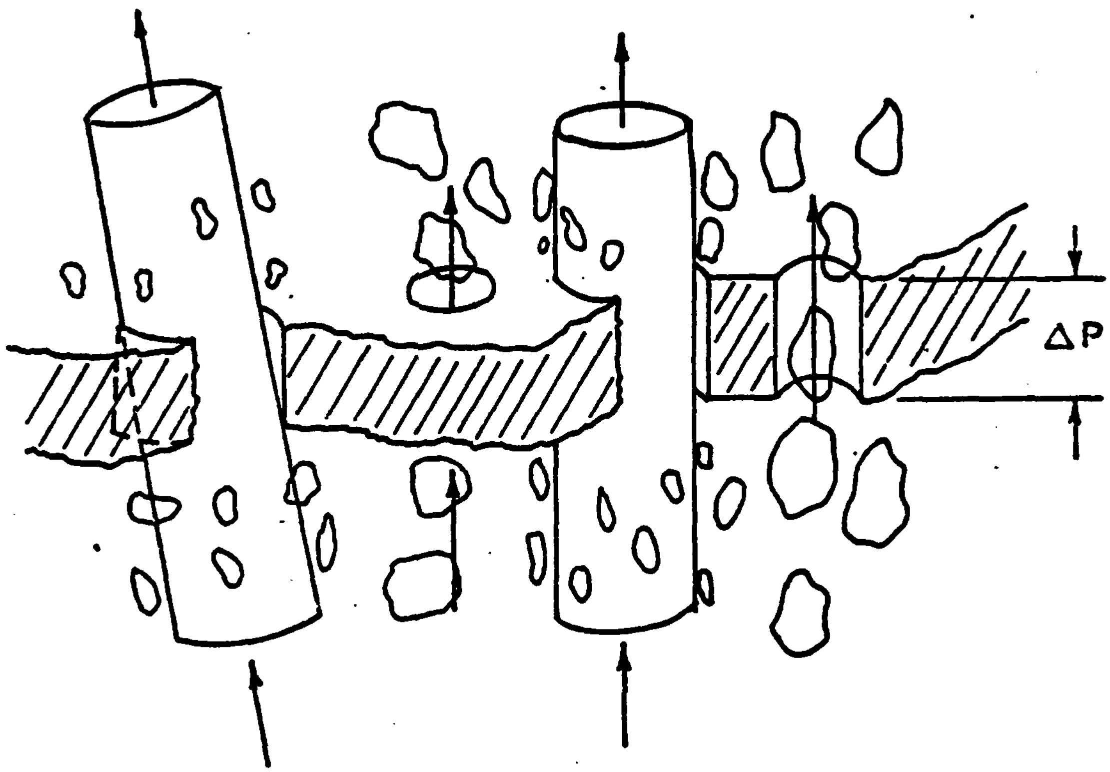
$$h\,(xy) = c\left(1 - \cos\left(\frac{x}{R}\right)\right) \tag{2}$$

for the line-contact condition of Figure 2(b),

$$\text{and}\quad h\,(xy) = c\left[1 - \left(2\,\frac{y}{L_y} - 1\right)\cos\left(\frac{x}{R}\right)\right] \tag{3}$$

for the two-point-contact condition of Figure 2(a).

(a)



(b)

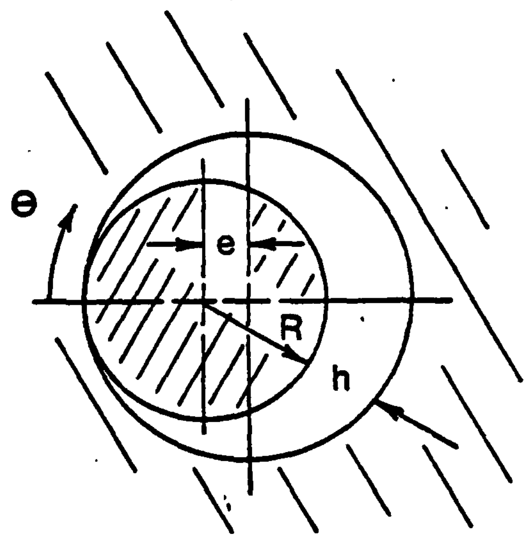


Figure 1

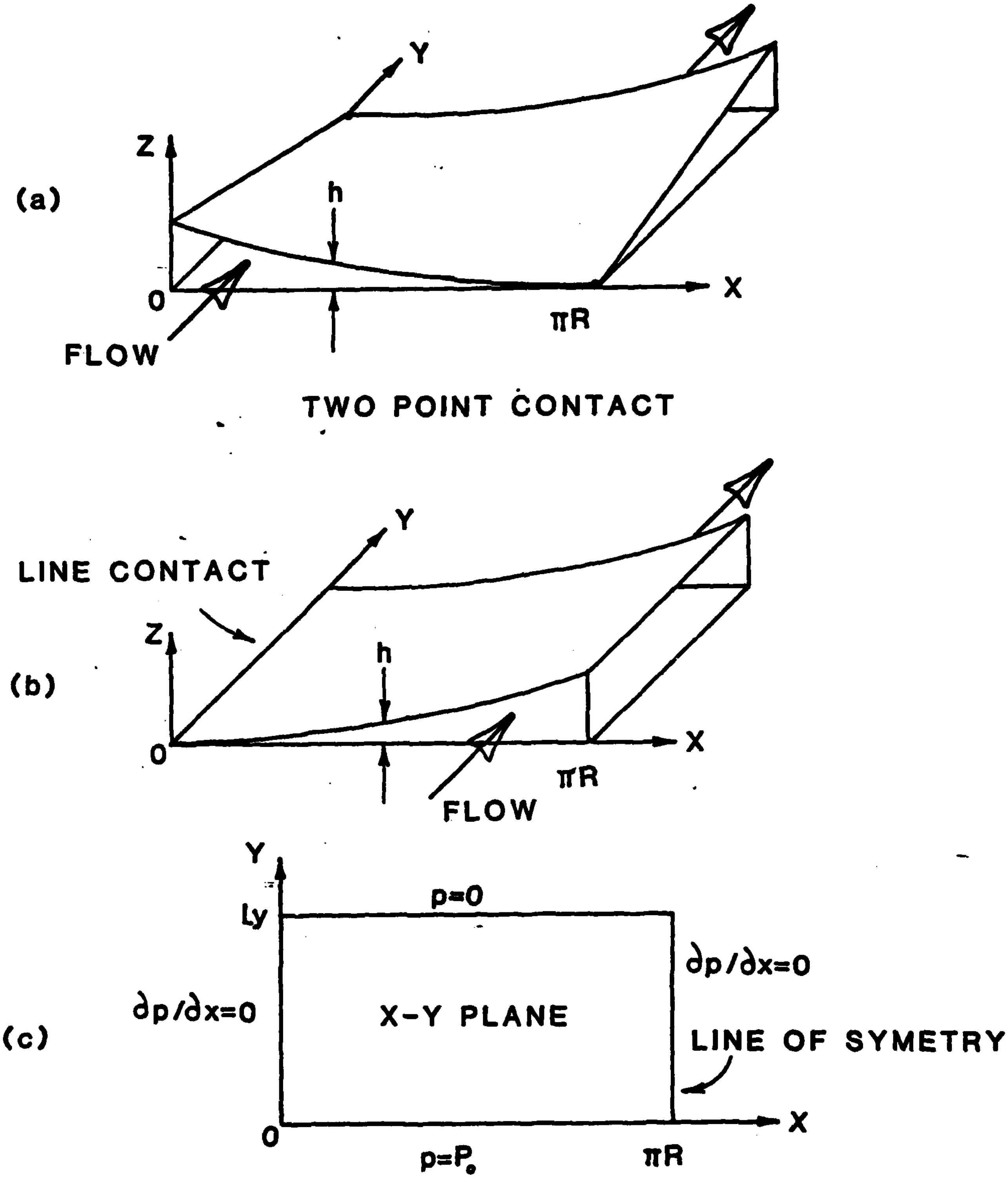

(a)
Y
Z
h
0
X
πR
FLOW
TWO POINT CONTACT
LINE CONTACT
Y
Z
(b)
h
0
X
πR
FLOW
Y
p=0
Ly
∂p/∂x=0
∂p/∂x=0
X-Y PLANE
(c)
LINE OF SYMETRY
0
X
p=Pₒ
πR

Figure 2

A. Subchannel Analysis

Viewing the flow field from the x-y plan as many individual subchannels along the y axis, the conservation equations can be formulated on each node of each subchannel. The distinct feature is that the forms of the conservation equations can be greatly simplified as compared with the conventional differential analysis, such that, efficient calculation can be achieved but with reasonable accuracy. As a result of the simplified model, the subchannel analysis is able to calculate the complicated two phase flow and boiling heat transfer effectively.

In the present analysis, the flow field is calculated on the x-y plan with the varying gap thickness h considered. The velocity and temperature in the analysis are the averaged values over the thickness h. To simplify the flow field calculation, it is assumed that the axial pressure gradient is identical at any point at a same y elevation [4].

1. Formulation

a. Axial Flow

At steady state two phase flow in the confined space, the pressure force is balanced by the frictional force. That means

$$- \Delta p = \Phi f_{\ell} \frac{\Delta y}{D_h} \frac{1}{2} \frac{G^2}{\rho_{\ell}} \tag{4}$$

where the f_{ℓ} is the friction factor of liquid flow with a same amount of mass flow rate, G is the mass flux, and Φ is the two phase friction multiplier. Define F_i as the mass flow rate over unit length in the x-y plan in subchannel i. This gives

$$F_i = \rho_\ell \bar{u}_i h_i \tag{5}$$

Also, the pressure gradient at the elevation j is the same for all the subchannels. Set that as

$$-\frac{\Delta p}{\Delta y} = C_j^2 \tag{6}$$

From the above three equations, the local axial mass flow becomes

$$F_{ij} = C_j h_{ij} \left[\frac{2\rho_\ell D_{hij}}{\Phi_{ij} f_{\ell ij}}\right]^{1/2} \tag{7}$$

where i denotes the subchannel, and j is the node notation in y direction as illustrated in Figure 3.

The sum of the axial flows should equal the total mass flow

$$\dot{m}_{tot} = \Delta x C_j \sum_i^n h_{ij} \left[\frac{2\rho_\ell D_{hij}}{\Phi_{ij} f_{\ell ij}}\right]^{1/2} \tag{8}$$

Assuming the two phase flow in the narrow gaps as homogeneous, the frictional multiplier can be described as

$$\begin{aligned} \Phi &= 1.0 \quad \text{if } x \leq 0 \\ &= \left(\frac{\rho_f}{\rho}\right) \quad \text{if } x > 0 \end{aligned} \tag{9}$$

where x is the local quality to be evaluated from energy balance. Better forms can be used for Φ when the experimental information becomes available from our tests.

b. Cross Flow

The cross flow between subchannels is calculated from mass conservation. Following the Figure 3(b), the mass conservation gives

$$F_e \Delta y = F_w \Delta y + F_s \Delta x - F_n \Delta x \tag{10}$$

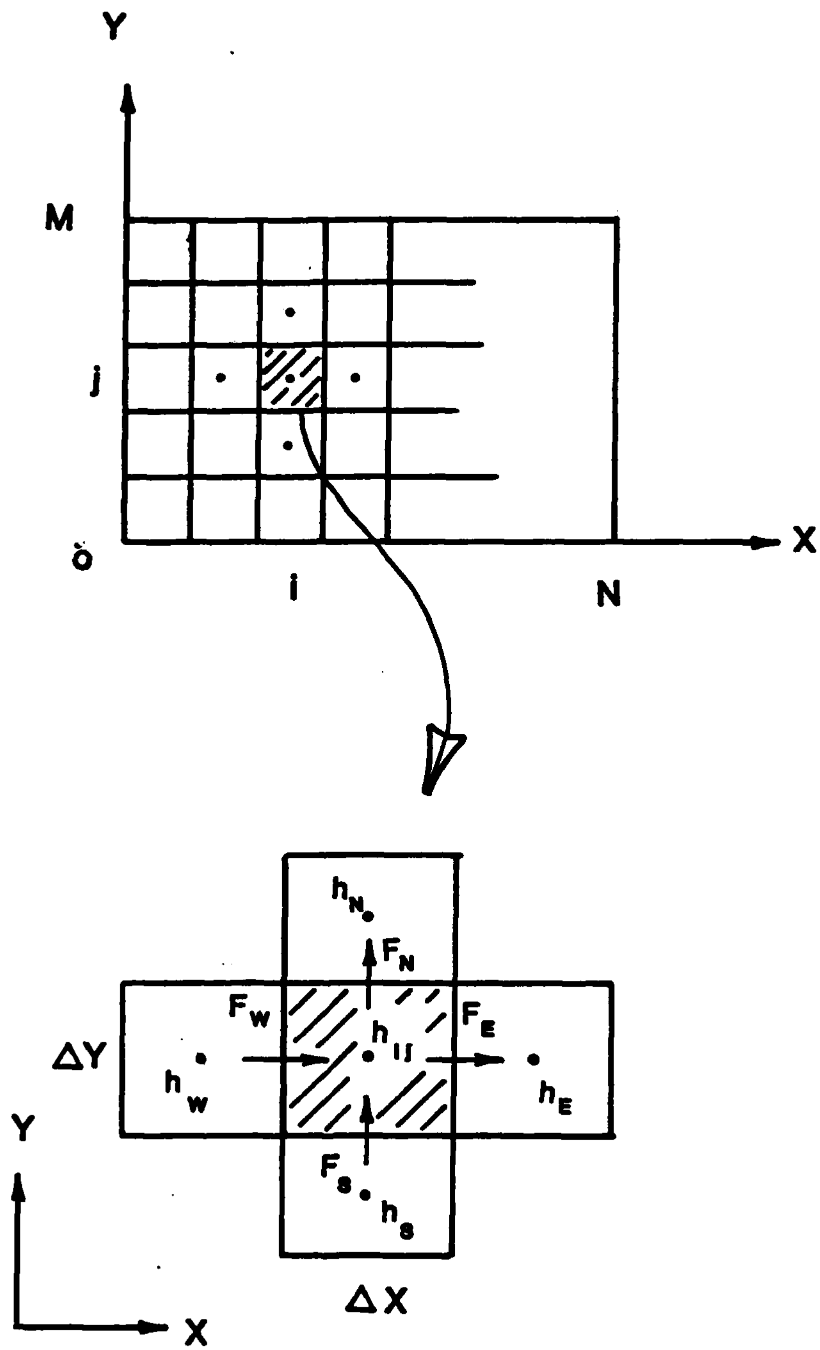


Figure 3

The F_s and F_n are known from the axial flow calculations. The boundary condition of symmetry at x = o gives

$$F_w \Big|_{x = o} = o \tag{11}$$

Therefore, all the cross flows at each subchannel can be evaluated easily starting from the locations x = o to x = ¶R.

c. Energy Balance

Following the illustration of Figure 3(b), the energy balance for a control volume ij can be written as

$$F_s \Delta x H_s + F_w \Delta y H_w + q_{ij} \Delta x \Delta y = F_e \Delta y \, H_{ij} + F_n \Delta x H_{ij} \tag{12}$$

where H is the enthalpy and q_{ij} is the heat flux from walls.

The F_n can be evaluated from equation (10)

$$F_n \Delta x = F_w \Delta y + F_s \Delta x - F_e \Delta y \tag{13}$$

Substituting the F_n into equation (12) the H_{ij} can be derived

$$\begin{aligned} H_{ij} &= H_s + q_{ij} \, \Delta y / F_s \\ &\quad + \frac{F_w \Delta y}{F_s \Delta x} \left(H_w - H_{ij}\right) \\ &\quad + \frac{F_e \Delta y}{F_s \Delta x} \left(H_e - H_{ij}\right) \end{aligned} \tag{14}$$

The third term at right hand side of this equation will be set to zero if F_w is flowing in the negative x direction; the fourth term will be set to zero if F_e is flowing in positive x direction.

With the known enthalpy the local quality can be evaluated.

$$x_{ij} = (H_{ij} - \Delta H_{sub}) / H_{fg} \quad \text{for } 0 \leq x \leq 1 \tag{15}$$

where ΔH_{sub} is the enthalpy of inlet subcooling.

2. Numerical Method

The calculational procedure is shown in Figures 4 and 5. Iteration is performed to achieve the convergence of flow rate on the line j, the pressure drop across the confined space, and the local quality at all the nodes. Due to the simplicity of the model and formulations, the convergence is rapid.

3. Results

A computer program has been developed for the subchannel analysis of two phase boiling in confined space. The condition of line-contact (refer to Figure 2(b)) is calculated for two phase boiling with dry-out.

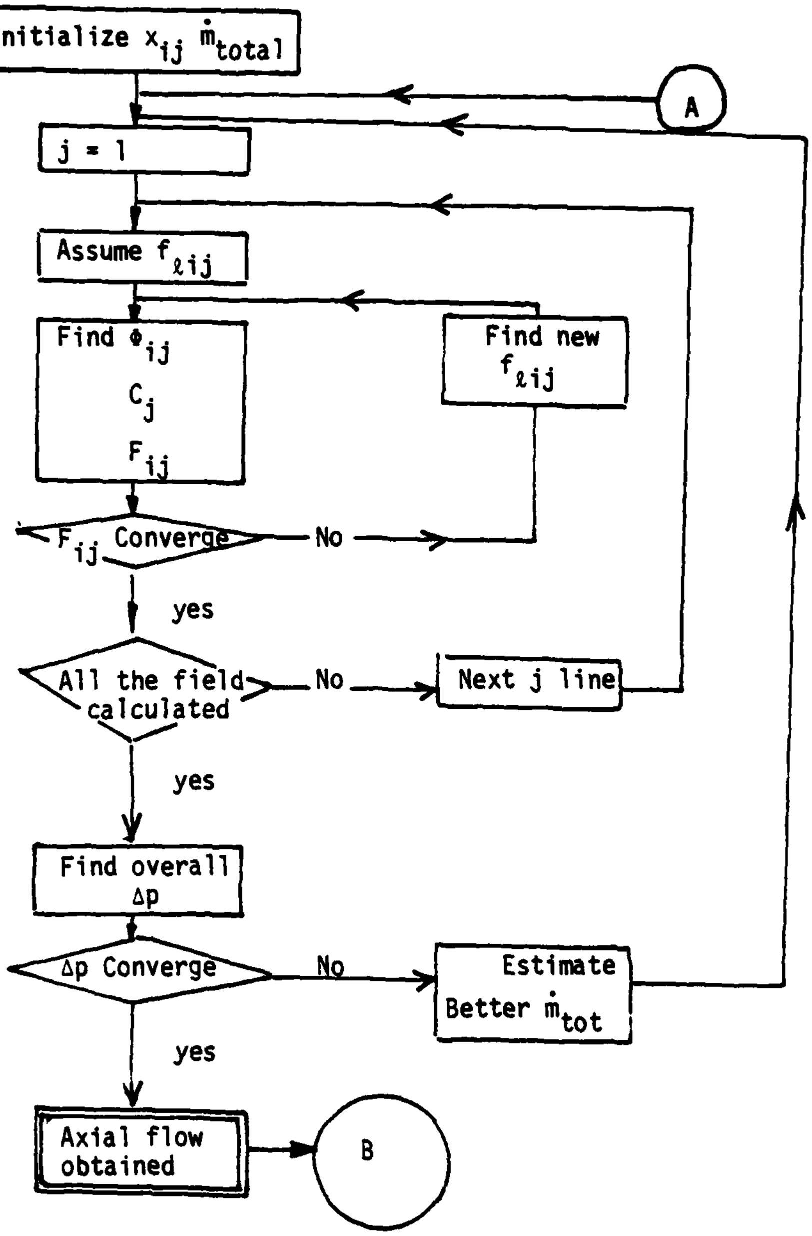

Initialize x_{ij} $\dot{m}_{total}$
A
j = 1
Assume $f_{\ell ij}$
Find Φ_{ij}
C_j
F_{ij}
Find new
$f_{\ell ij}$
F_{ij} Converge
No
yes
All the field calculated
No
Next j line
yes
Find overall
Δp
Δp Converge
No
Estimate
Better $\dot{m}_{tot}$
yes
Axial flow
obtained
B

Figure 4

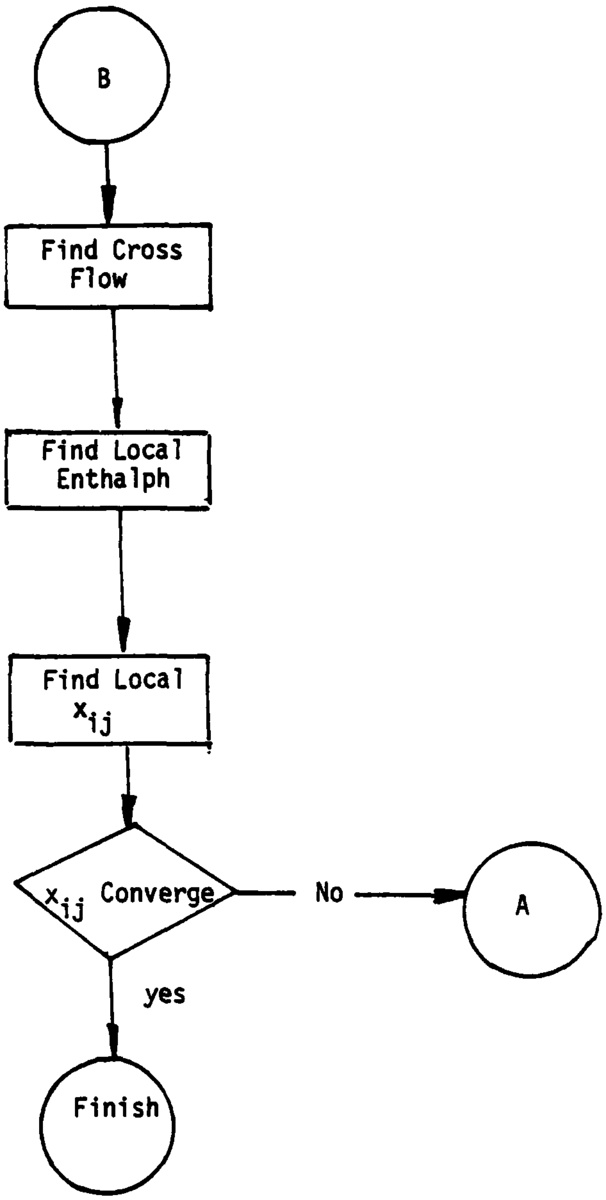
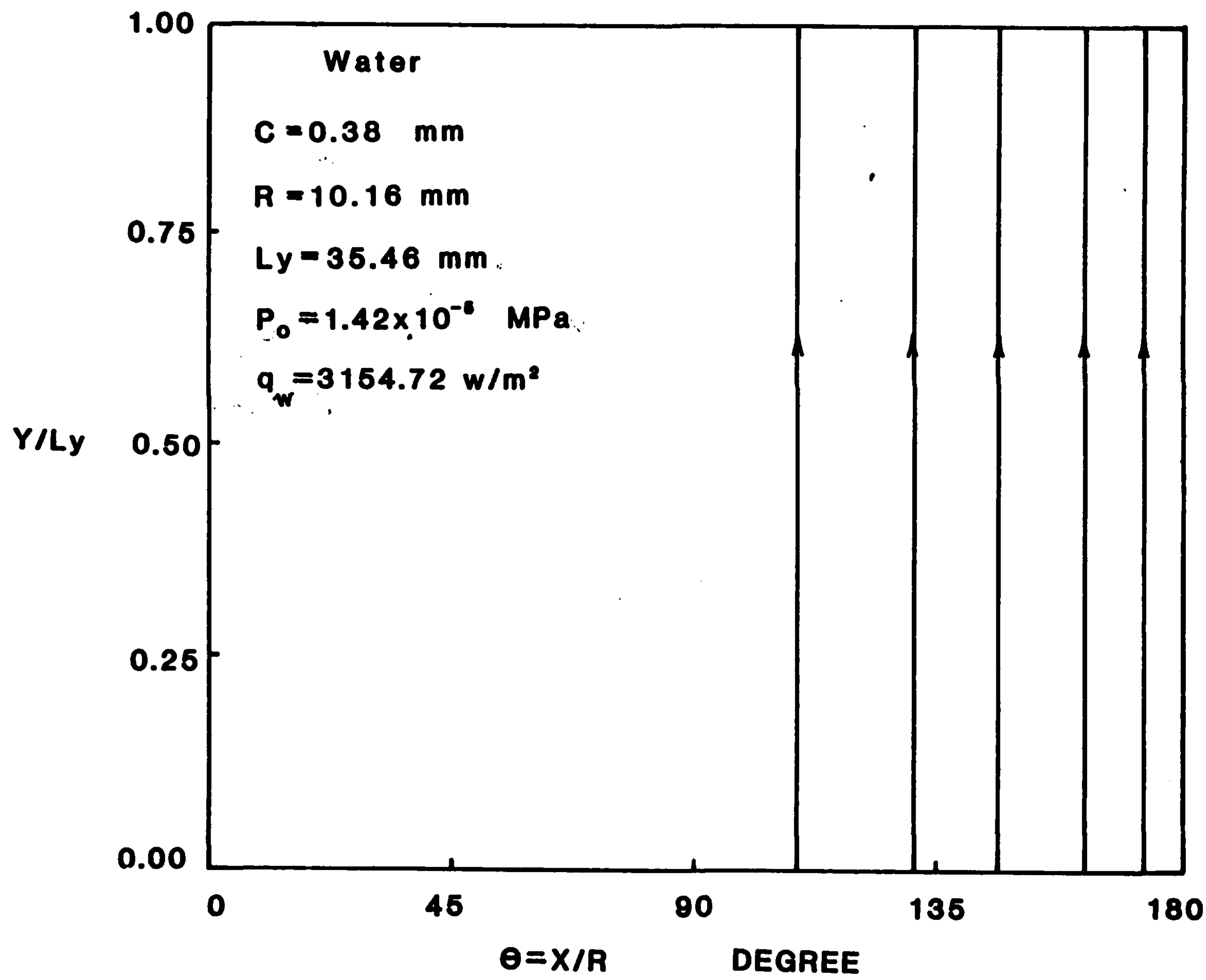

B
Find Cross Flow
Find Local Enthalph
Find Local x_{ij}
x_{ij} Converge
No
A
yes
Finish

Figure 5

The two phase flow and boiling are calculated for condition of line contact at constant heat flux condition. The stream line of the fluid flow is shown in Figure 6. Although dryout appears at locations near the line contact, since the gaps there is small and the dryout pattern is almost parallel to *y* axis, the stream lines are generally not deviated appreciably from a straight line.

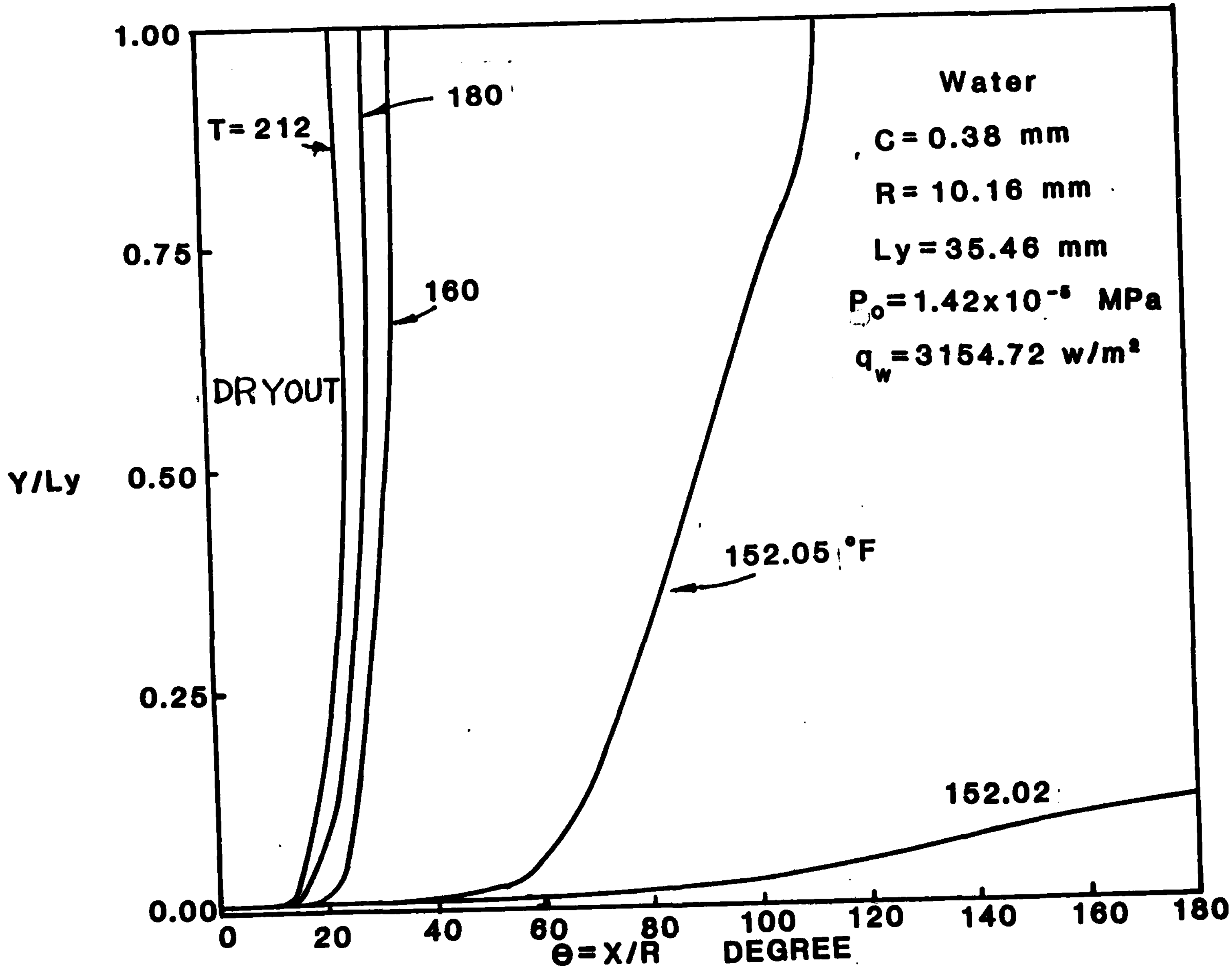
The constant temperature contour and the dryout pattern at the line contact condition is shown in Figure 7. At the location near 180°, the gap is large, the flow is strong, and the fluid temperature does not rise much from the inlet value. Near the line of contact, the fluid temperature rises drastically and reaches dryout in short distance of flow. The dryout region expands rapidly then becomes parallel to the *y* axis. It is believed that at higher system pressures, the boundary between liquid region and dryout region will be a distinct zone of two phase flow.

Finally, it is important to point out that the above analysis contains many assumptions. The justfication of these assumptions, or the future modification of the model, will be based upon the comparison with experimental data. However, when the model becomes mature, the boiling in confined space can be predicted for any fluid at any working conditions.



Water
C = 0.38 mm
R = 10.16 mm
Ly = 35.46 mm
Po = 1.42x10^-5 MPa
qw = 3154.72 w/m²
Y/Ly
1.00
0.75
0.50
0.25
0.00
0
45
90
135
180
Θ=X/R DEGREE

Figure 6


Water
C = 0.38 mm
R = 10.16 mm
Ly = 35.46 mm
P_o = 1.42x10^{-5} MPa
q_w = 3154.72 w/m^2
T= 212
180
160
DRYOUT
152.05 °F
152.02
Y/Ly
1.00
0.75
0.50
0.25
0.00
0
20
40
60
80
100
120
140
160
180
θ = X/R DEGREE

B. Differential Analysis

1. Formulation

Precised conservation equations can be established in the differential form for the fluid. Due to the nature of the narrow flow channel, approximations following the lubrication theory will be adopted. Although the present analysis is limited to the single phase fluid flow, the results could be used as a reference to validate the more approximated sub-channel analysis which has been used by us to study the two phase boiling phenomenon in the confined space.

If the fluid is incompressible, the mass conservation equation becomes

$$\frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} + \frac{\partial w}{\partial z} = 0 \qquad (16)$$

or

$$\frac{\partial w}{\partial z} = -\frac{\partial u}{\partial x} - \frac{\partial v}{\partial y} \qquad (17)$$

The momentum equations can be greatly simplified according to the order of magnitude analysis considering the narrow gaps. At laminar flow they become

$$\frac{\partial p}{\partial x} = \mu \frac{\partial^2 u}{\partial z^2} \qquad (18)$$

$$\frac{\partial p}{\partial y} = \mu \frac{\partial^2 v}{\partial z^2} \qquad (19)$$

$$\frac{\partial p}{\partial z} = 0 \qquad (20)$$

Parabolic velocity profiles can be obtained through the integration of equations (18) and (19). Then, the velocity profiles can be substituted into equation (17) and integrate

the dz from 0 to h. The resulting equation becomes

$$\frac{\partial}{\partial x}\left(h^3 \frac{\partial p}{\partial x}\right) + \frac{\partial}{\partial y}\left(h^3 \frac{\partial p}{\partial y}\right) = 0 \qquad (21)$$

which is the Reynolds equation of lubrication theory.

In this equation, both h and p are function of space. The boundary condition for forced flow is

$$p\ (x,o) = P_0 \qquad (22)$$

$$p\ (x,L_y) = o \qquad (23)$$

$$\frac{\partial p}{\partial x}\ (o,y) = o \qquad (24)$$

$$\frac{\partial p}{\partial x}\ (\pi R,y) = o \qquad (25)$$

Once the pressure field p(xy) has been solved, the "averaged" fluid velocity in the gap can be evaluated.

$$\bar{u} = -\frac{1}{12\mu} \frac{\partial p}{\partial x} h^2 \qquad (26)$$

$$\bar{v} = -\frac{1}{12\mu} \frac{\partial p}{\partial y} h^2 \qquad (27)$$

This set of equations can be non-dimensionalized by using

$$X = \frac{x}{\pi R} \qquad (28)$$

$$Y = \frac{y}{L_y} \qquad (29)$$

$$\bar{h} = \frac{h}{c} \qquad (30)$$

$$P = \frac{p}{P_0} \qquad (31)$$

$$U = \frac{\bar{u}\mu\ \pi R}{P_0 c^2} \qquad (32)$$

$$V = \frac{\bar{v}\mu L_y}{P_0 c^2} \qquad (33)$$

and

$$S = \frac{L_y}{\pi R} \qquad (34)$$

We can further define

$$Pe = Re \cdot Pr = VoLy/\alpha \tag{35}$$

where

$$V_o = PoC^2/\mu Ly$$

Finally the governing equation for pressure field becomes

$$\frac{\partial}{\partial X}\left(\bar{h}^3 \frac{\partial P}{\partial X}\right) + \frac{\partial}{\partial Y}\left(\bar{h}^3 \frac{\partial P}{\partial Y}\right) = 0 \tag{36}$$

where $\bar{h} = [1 + (1-2Y)]\cos(\pi x)$ for 2-point contact with boundary conditions (37)

$$P(X,0) = 1 \tag{38}$$

$$P(X,1) = 0 \tag{39}$$

$$\frac{\partial P}{\partial X}(0,Y) = 0 \tag{40}$$

$$\frac{\partial P}{\partial Y}(1,Y) = 0 \tag{41}$$

From the solved pressure field, the velocity field can be found.

$$U = -\frac{1}{12}\bar{h}^2 \frac{\partial P}{\partial X} \tag{42}$$

$$V = -\frac{1}{12}\bar{h}^2 \frac{\partial P}{\partial Y} \tag{43}$$

The energy equation can be written in terms of gap averaged conditions. The final energy equation becomes

$$S^2 U \frac{\partial \bar{T}}{\partial X} + V \frac{\partial \bar{T}}{\partial Y} = \frac{1}{\bar{h}\, Pe^*} \tag{44}$$

(This equation only for constant heat flux.)

where $Pe^* = Pe \cdot \left(\frac{C}{Ly}\right)^2$

where

$$\bar{T} = \frac{k(T-To)}{q_w C} \tag{45}$$

with initial conditions

$$\bar{T} = 0 \quad \text{at } Y = 0 \tag{46}$$

$$\frac{\partial \bar{T}}{\partial X} = 0 \quad \text{at } X = 0 \tag{47}$$

2. Numerical Method

The above formulation have been written into finite difference form to solve for the pressure field. Different x and y increments are allowed in the program. The whole equations are solved by Alternative Directional Implicite method [5] to speed up the convergence of the solution. Due to the special *boundary condition* at the line $x = o$ and $x = \pi R$ (equations (24) and (25)), the implicite calculation is not performed for these two lines during the sweep of A.D.I. method. During the iteration, the implicite equations on a same line are arranged in the form of tridiagonal matrix. Then Gauss elimination method is performed for the solution of the pressure field.

3. Results

It is obvious that the flow field of single phase flow at the condition of line contact will have straight stream line along the y direction. For a same pressure difference across the tube support plate the overall flow rate of this differential analysis is about the same value as that of the subchannel analysis. This confirms the compatibility of these two approaches.

The typical flow field of the two-point-contact condition is shown in the Figure 8. The arrows indicate the flow direction and magnitude. The curves show the axial component. They possess the feature of symmetry and indicate the turning of the flow.

The temperature field of two-point-contact condition at constant heat flux condition is shown in Figure 9 for the case of S = 1.11. The contacting points show high fluid temperature. Near the contact point of (1.0), hot fluid extends to downstream, but diminishes gradually. Near the point (0.1) the hot region is rather localized. Viewing the overall temperature distribution, it is clear that the dry-out phenomena will not be as severe as that of the line-contact conditions.

III. EXPERIMENT

A. Test Loop

In the past year, much effort has been devoted to the design and construction of the test loop. At the present time, the loop is completed. Test runs indicate that the design requirements are fully satisfied.

The schematic of the loop is shown in Figure 10 with the loop charactersistics summarized in the Table 1.

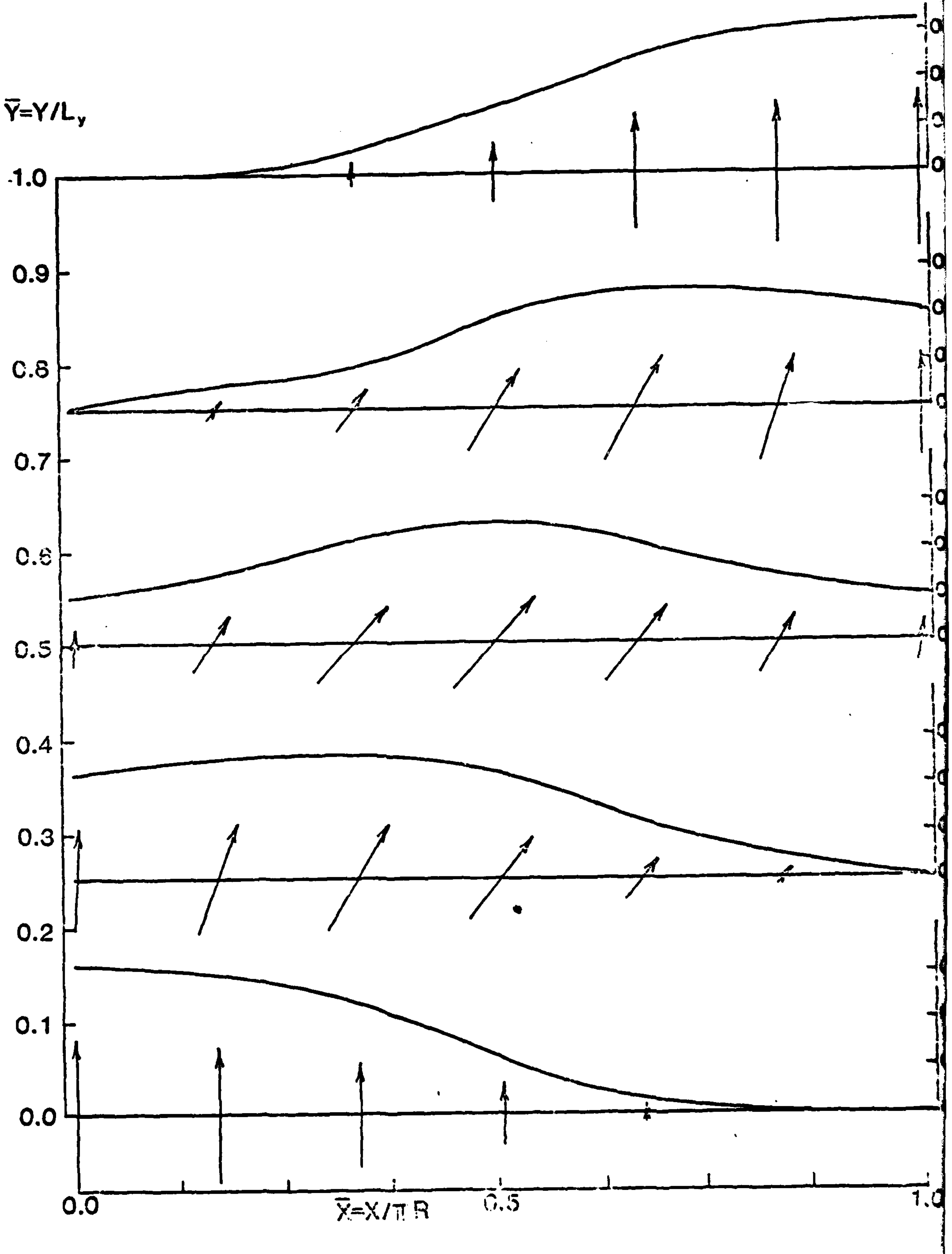


Fig. 8 The mass flow rate in two point contact case

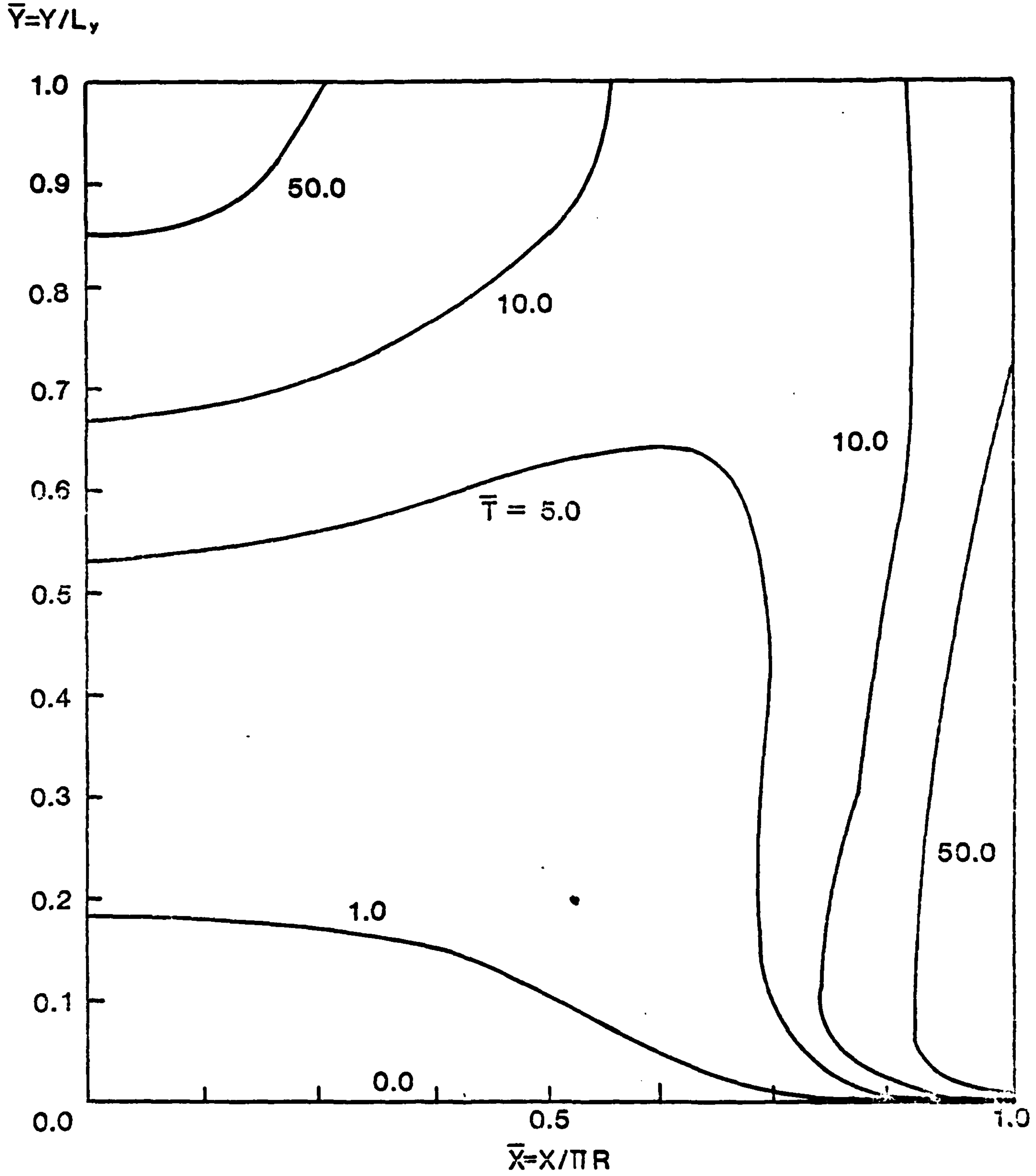


Fig.9 The non-dimensional temperature distribution at constant heat flux in two point contact case

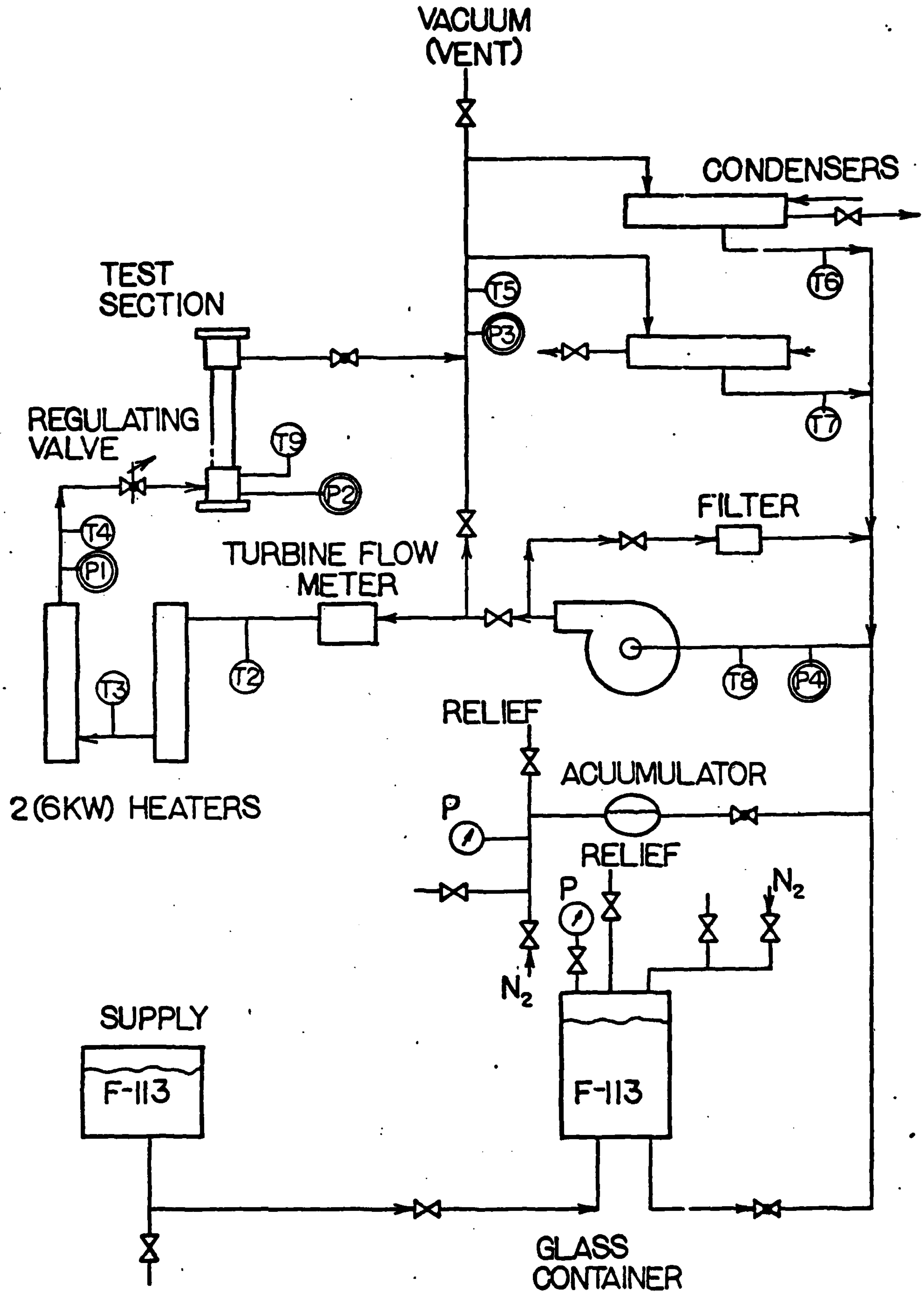

VACUUM
(VENT)
CONDENSERS
TEST
SECTION
T5
P3
T6
T7
REGULATING
VALVE
T9
P2
FILTER
T4
P1
TURBINE FLOW
METER
T3
T2
T8
P4
RELIEF
ACUUMULATOR
2 (6KW) HEATERS
P
RELIEF
P
N_2
N_2
SUPPLY
F-113
F-113
GLASS
CONTAINER

Figure 10

TABLE 1. LOOP CHARACTERISTICS

Working fluid	F - 113
System Pressure	0.1 ~ 1.0 MPa
Volume flow rate	0 ~ 68 LPm
Preheater Power	12 KW

In order to perform boiling experiments at constant pressures, an accumulator is attached to the loop to absorb the volume expansion of the two phase fluid during the tests. Large flow by-pass is installed parallel to the test section to maintain the flow stability in the test section.

The instrumentation system has also been established. The instrumentation rack includes the digital temperature displays, turbine flow meter readings, and the strip chart recorder for loop temperatures. The Hesse pressure gauges are mounted on the loop.

B. Test Section

The test sections have been designed and they are under fabrication. The schematic of pool boiling set up for studying the tube sheet crevices, is shown in Figure 11. The test section is heated directly with an internal moveable thermocouple for measuring the inner wall temperature. Through heat conduction calculation, the outer wall temperature can be calculated and then the local heat transfer coefficient is obtained. The pool can be pressurized to about 0.6 MPa working with F - 113 or water.

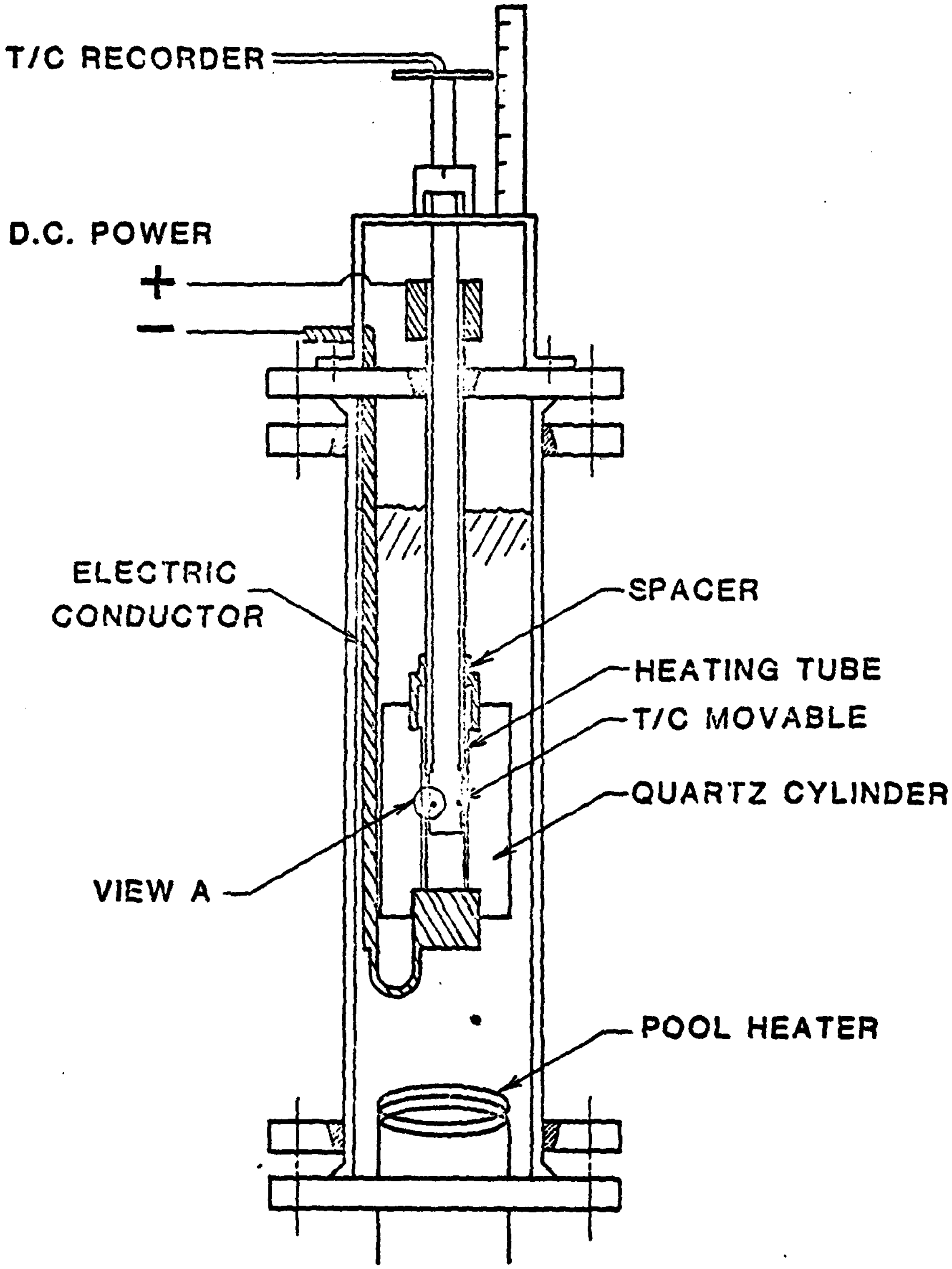

T/C RECORDER
D.C. POWER
+
−
ELECTRIC CONDUCTOR
SPACER
HEATING TUBE
T/C MOVABLE
QUARTZ CYLINDER
VIEW A
POOL HEATER

Fig. 11

The schematic of forced convective test section is shown in Figures 12 and 13. The quartz test section is changeable (one of 2.5 cm length, and three of 7.5 cm length with diffetent I.D.'s).

When the tube is heated under a constant heat flux conditon, the study of concentric annulus will be performed. The test sections can also be heated under constant temperature condition using a heat pipe for uniform temperature control. At this condition, the experiments of line-contact eccentric annulus will be performed and the dryout pattern will be studied. The dryout information will be compared with the prediction of our subchannel analysis.

IV. WORK TO BE DONE

The heat transfer analysis will be continued for the conditions of constant wall temperature conditions. Systematic study of the flow and heat transfer processes under various conditions will be studied through these analysis to achieve a better understanding of the boiling phenomena in confined space. Experimental research will be the major effort in the coming year and the data will be used to validate the subchannel analysis.

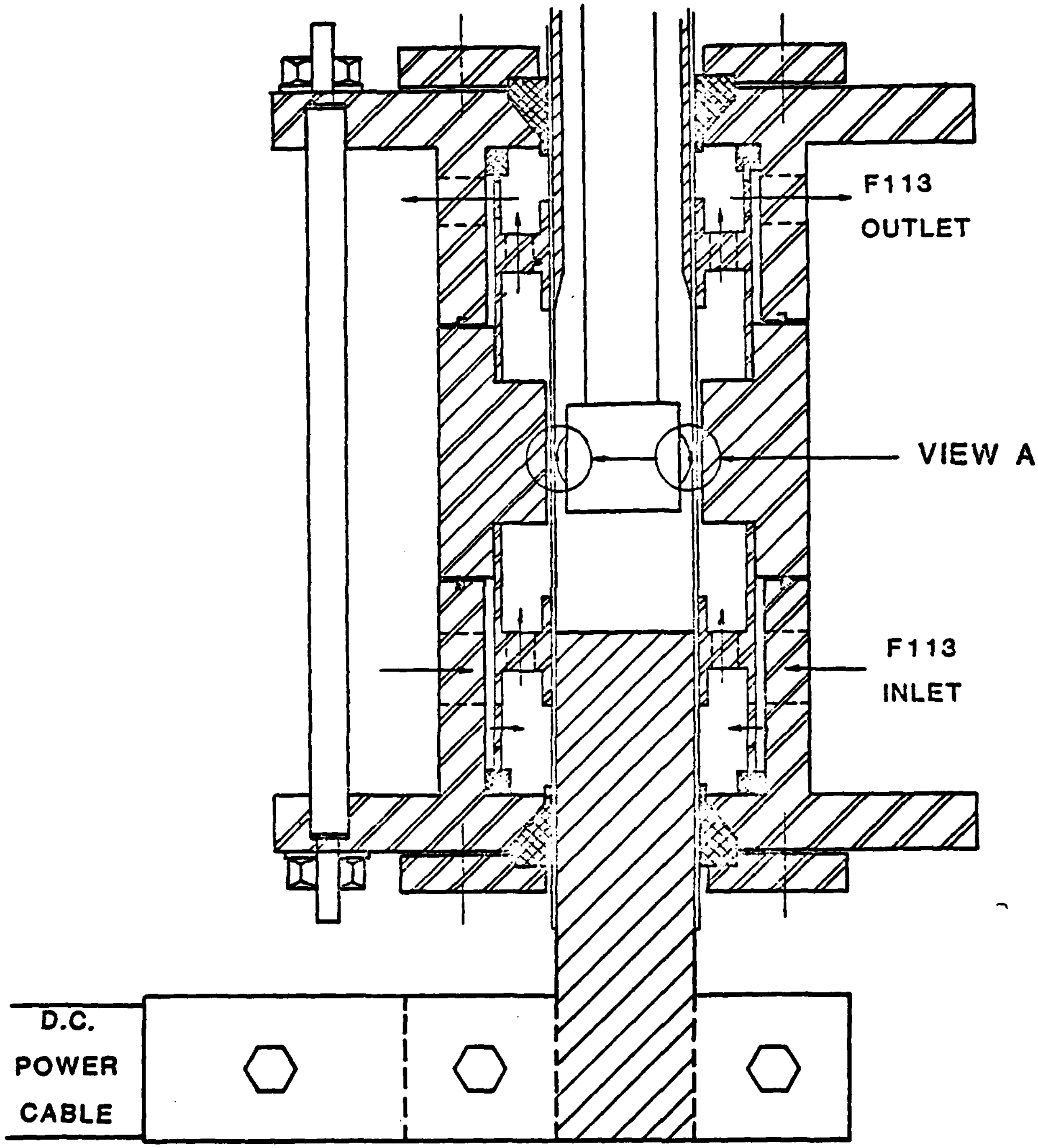
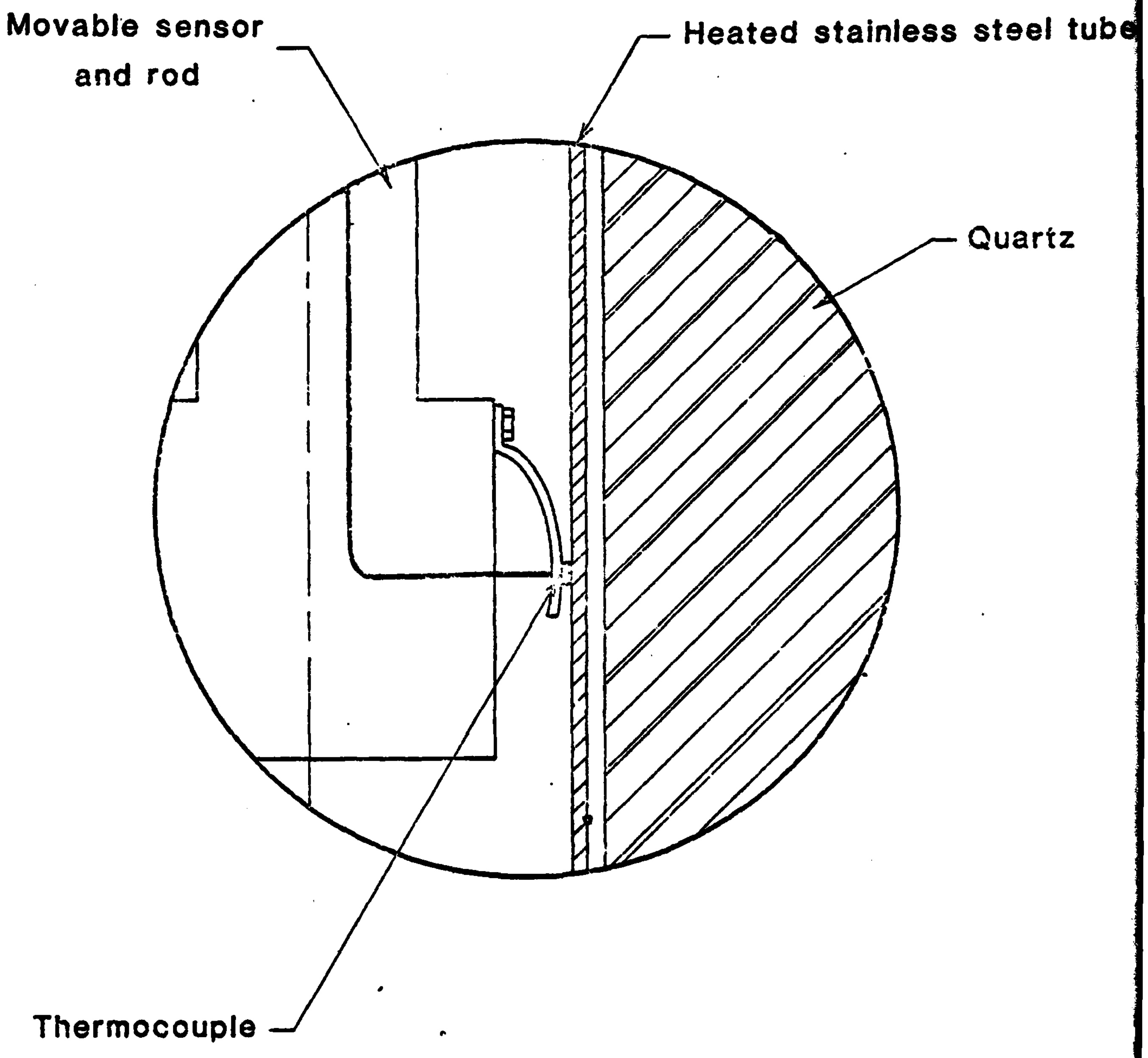


Fig. 12


Movable sensor
and rod
Heated stainless steel tube
Quartz
Thermocouple

VIEW A

Fig. 13

V. NOMENCLATURE

c	average gap thickness of the confined space
c_j^2	the pressure gradient at location j
D_h	hydraulic diameter
e	eccentricity
f	frictional factor
F	mass flow rate over unit length in x-y plan
G	mass flux
h	local gap thickness of the confined space
$\bar{h}$	non-dimensionlyzed gap thickness, h/c
H	enthalpy
H_{fg}	latent heat of vaporization
ΔH_{sub}	enthalpy of inlet subcooling
L_y	total length of the confined space
$\dot{m}_{tot}$	total mass flow
p	local pressure
P	non-dimensional pressure
P_o	the pressure drop across the confined space
q_w	wall heat flux
R	radius of the heated tube
S	aspect ratio of the confined space, $L_y/\pi R$
u	local axial velocity
$\bar{u}$	local axial velocity average over gap thickness
U	non-dimensional axial velocity
v	local velocity in x direction
$\bar{v}$	local velocity averaged over gap thickness

V	non-dimensional velocity in x direction
w	velocity in z direction
x	coordinate; or local two phase quality
X	non-dimensional x coordinate
y	axial coordinate
Y	non-dimensional y coordinate
z	coordinate

Greek Symbols

α	thermal diffusivity
ε	non-dimensional eccentricity, e/c
θ	angle
μ	viscosity of the liquid
ρ	fluid density
Φ	two phase frictional multiplier

VI. REFERENCES

1. Tong, L. S., Boiling Heat Transfer and Two-Phase Flow, John Wiley & Sons, 1965.

2. M. Jensen, A. Bergles, and P. Cooper, "Boiling Heat Transfer and Dryout in Restricted Annular Geometries", 16th National Heat Transfer Conference, Paper No. AICHE-14, 1976.

3. E. Ishibashi, and K. Nichikawa, "Saturated Boiling Heat Transfer in Narrow Spaces", Int. J. of Heat Mass Transfer, Vol. 12, pp. 863-94, 1969.

4. O. C. Jones, S. C. Yao, and R. E. Henry, "SIMPLE-2: A Computer Code for Calculation of Steady-State Thermal Behavior of Rod Bundles with Flow Sweeping", Nuclear Engineering and Design, Vol. 41, pp. 205-217, 1977.

5. B. Carnahan, H. Luther, and J. Wilkes, Applied Numerical Methods, John Wiley & Sons, Inc., 1969.

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